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(54) **CONTROL VALVE FOR A STIRLING ENGINE**

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F16K 31/02 (2006.01)

(52) **U.S. Cl.**
USPC **60/521**; 60/522; 60/525; 251/129.15

(58) **Field of Classification Search**
USPC 60/517, 521, 522, 525; 251/129.15,
251/129.2

See application file for complete search history.

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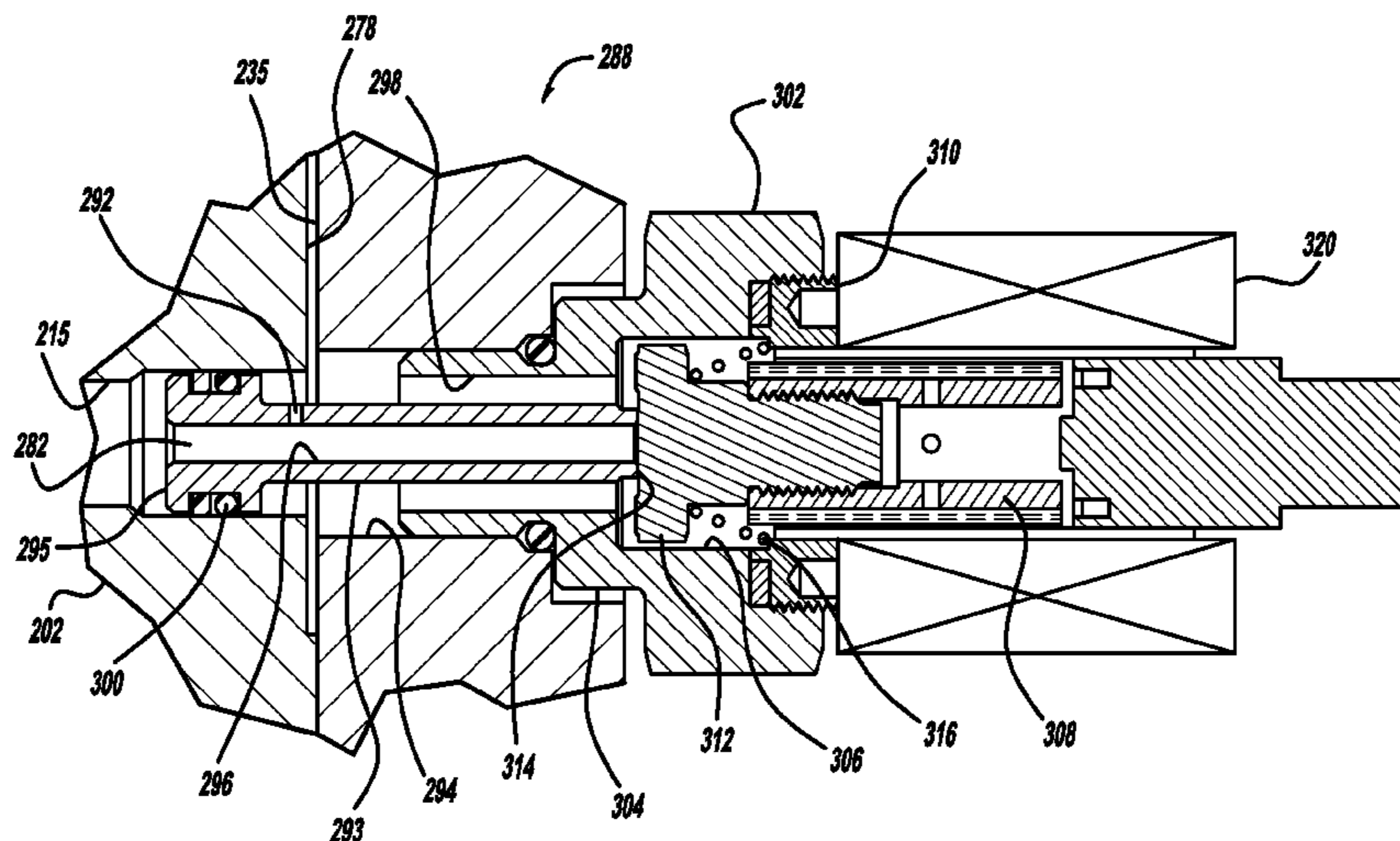
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(57) **ABSTRACT**

A control valve for a double acting type Stirling engine of the type having two or more piston assemblies reciprocating in bores with each of the piston assemblies separating isolated volumes of a working gas. The control valve provides the functions of a check valve which allows an internally defined volume within the engine to be held at a low pressure near the minimum pressure of the cycle volumes. The valve may further be actuated to unload the engine to provide low starting torque and to unload the engine in case of an engine or load malfunction. The control valve further provides a controlled leakage path for working fluid flowing between the cycle volumes and the minimum pressure volume which acts as part of a pressure balancing system for the Stirling engine which maintains the volume or mass of a working fluid in a balanced condition between the cycle volumes.

18 Claims, 12 Drawing Sheets



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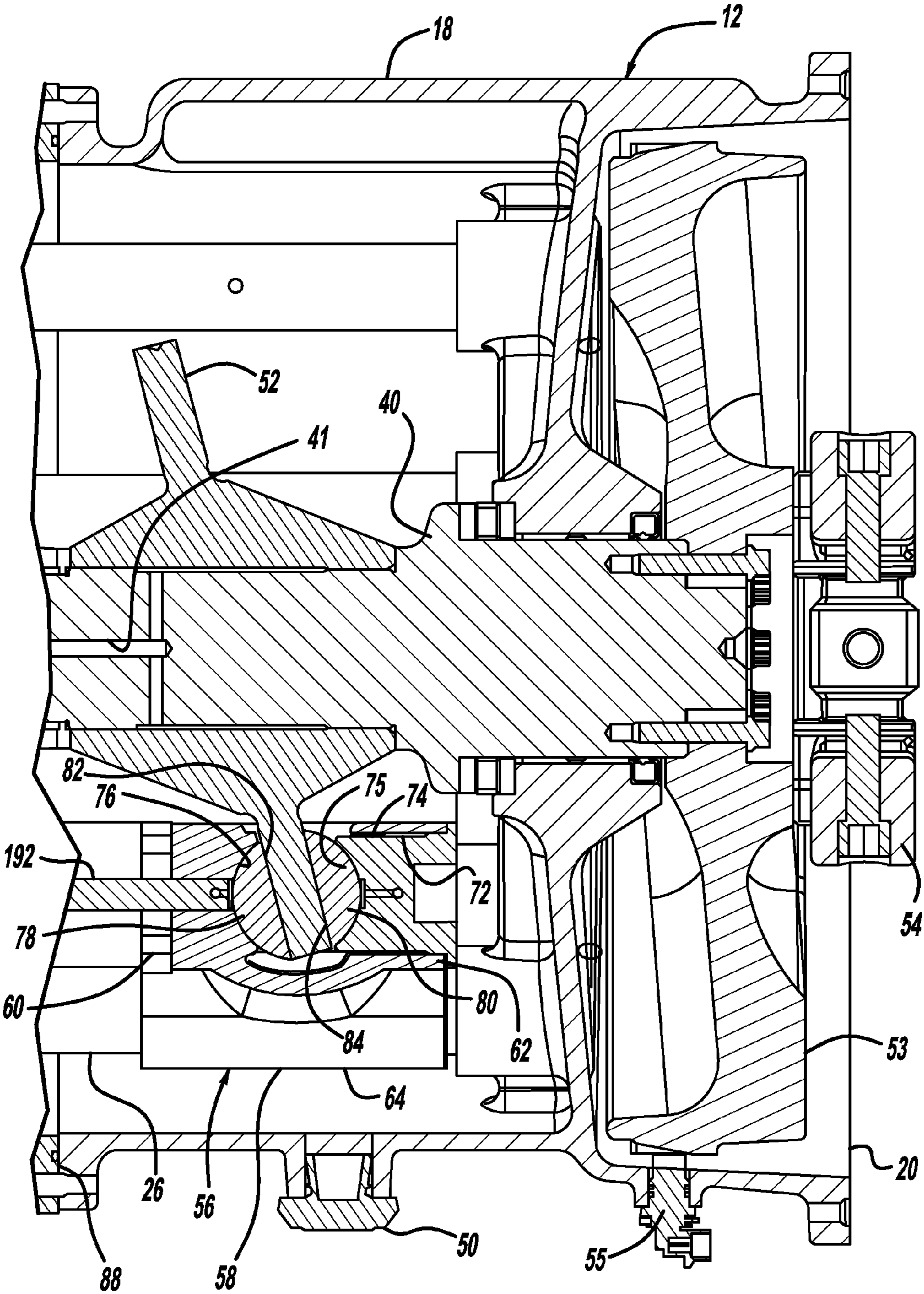
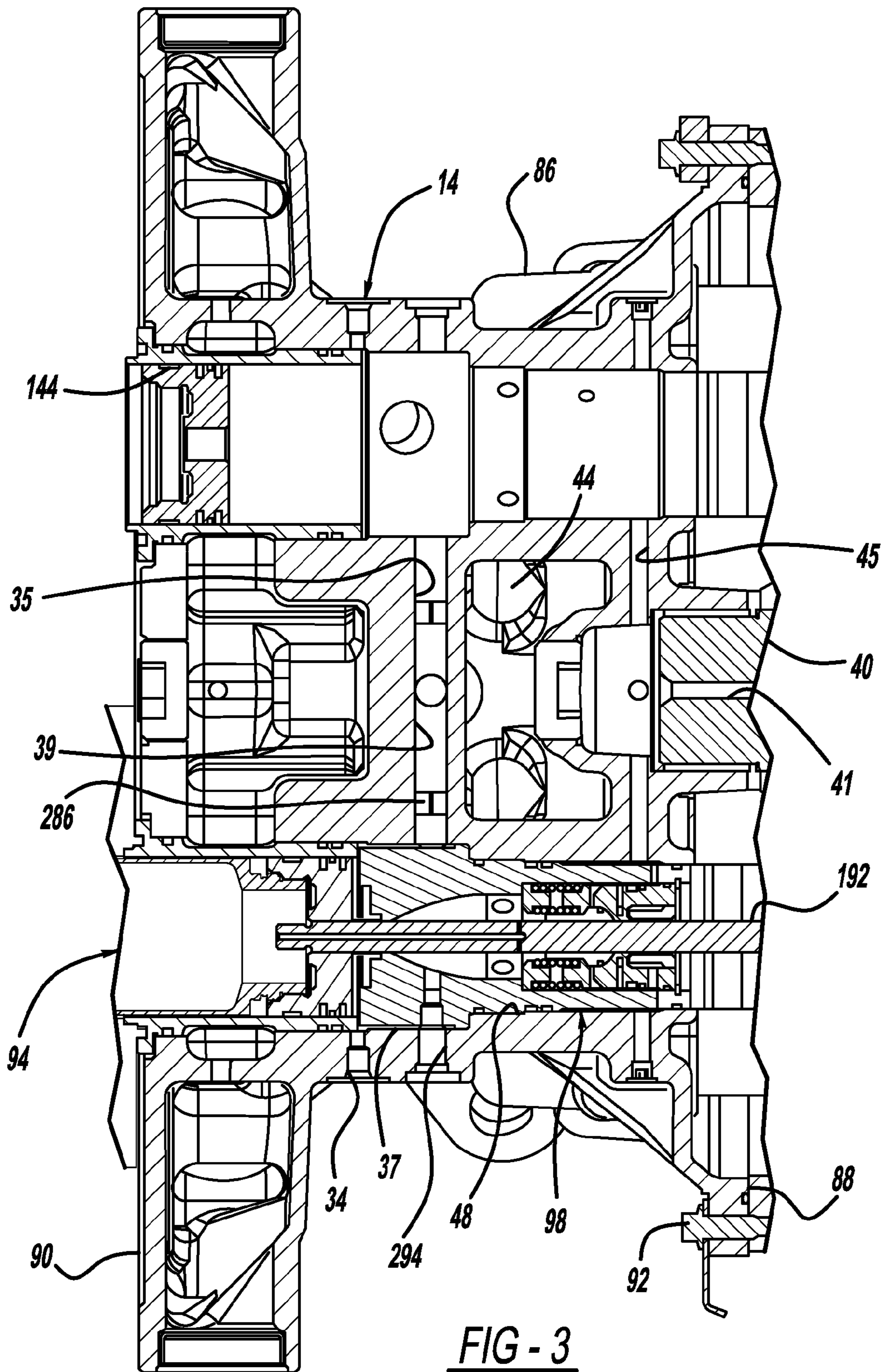
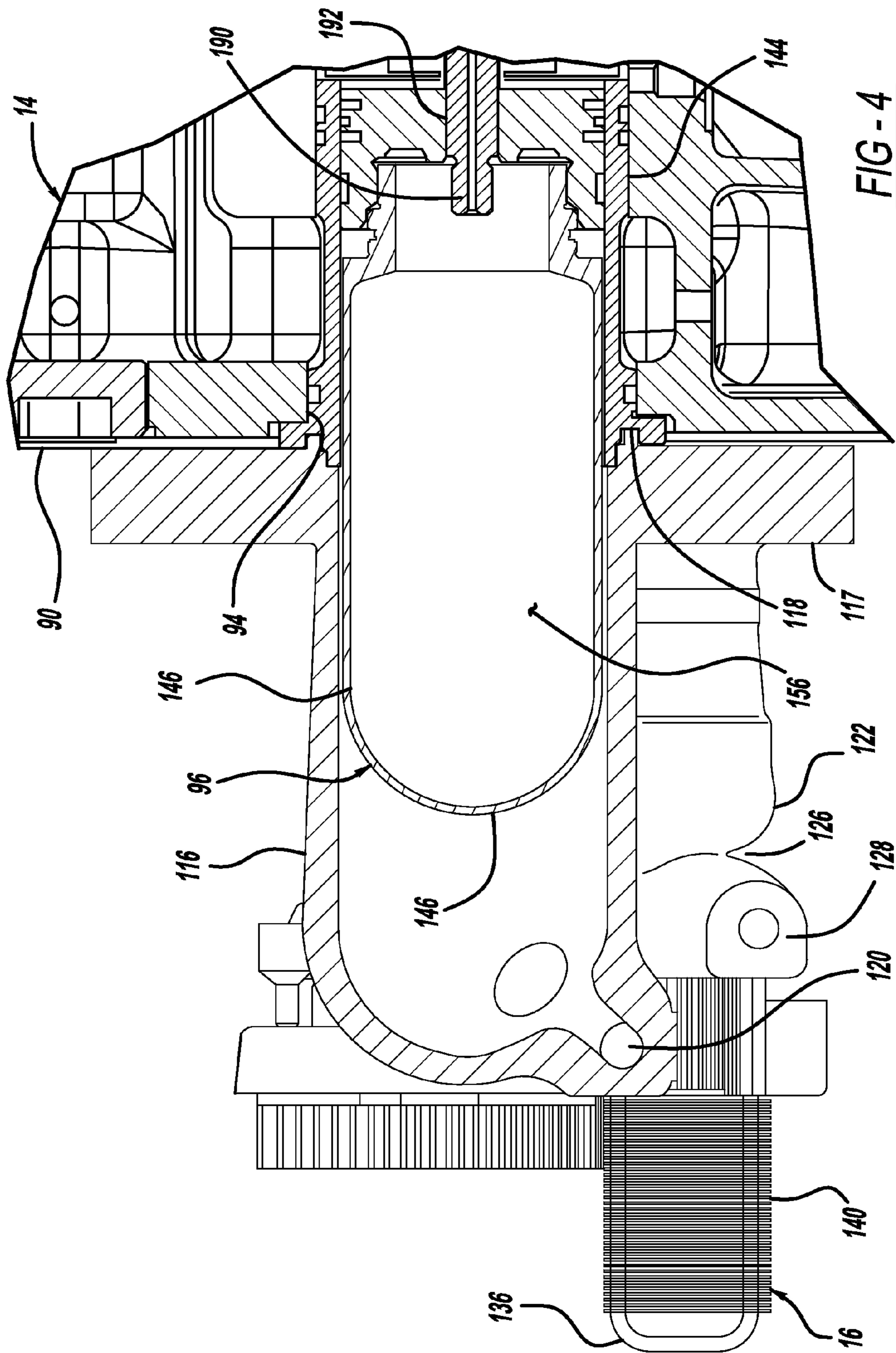


FIG - 2





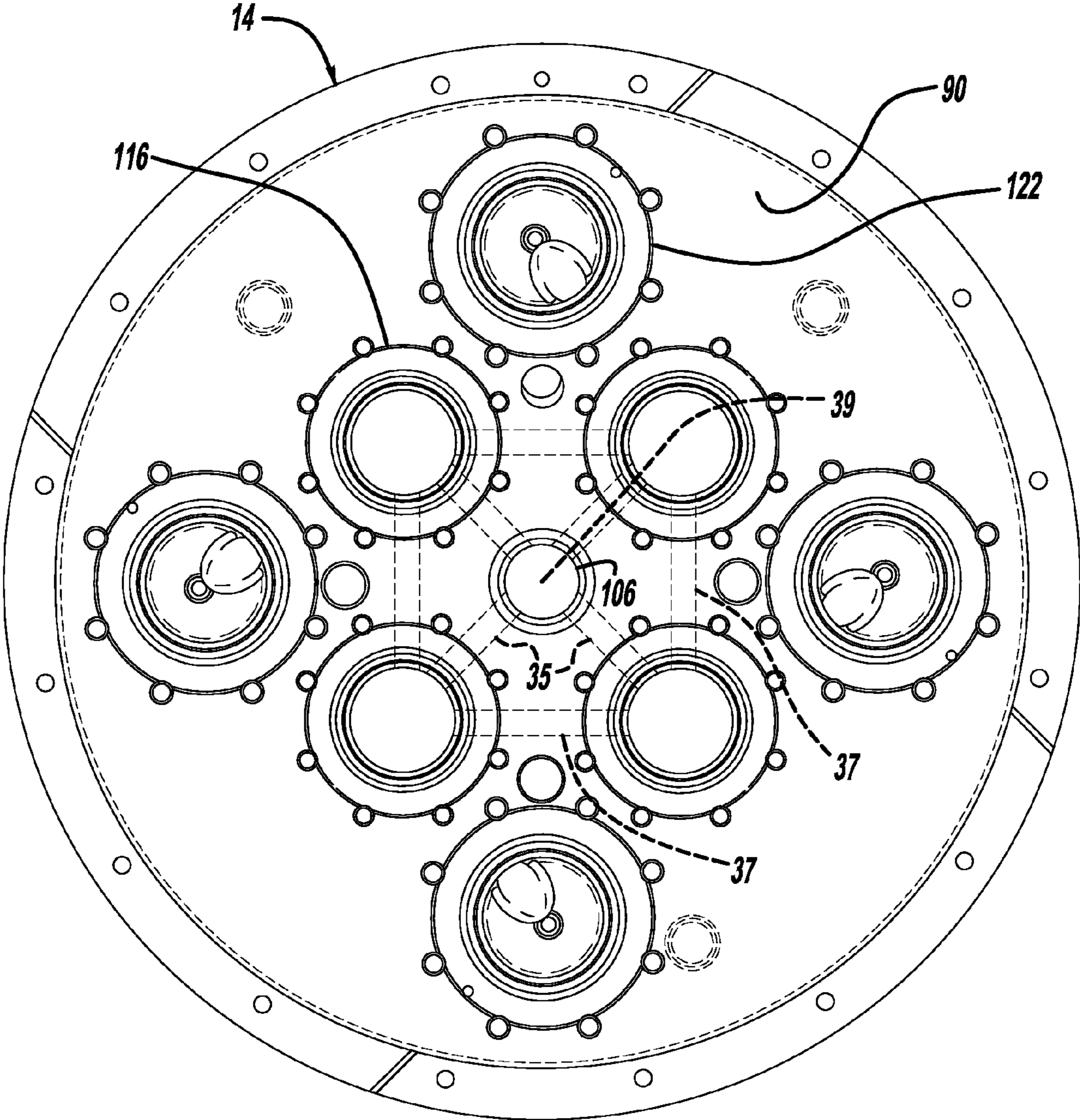


FIG - 5

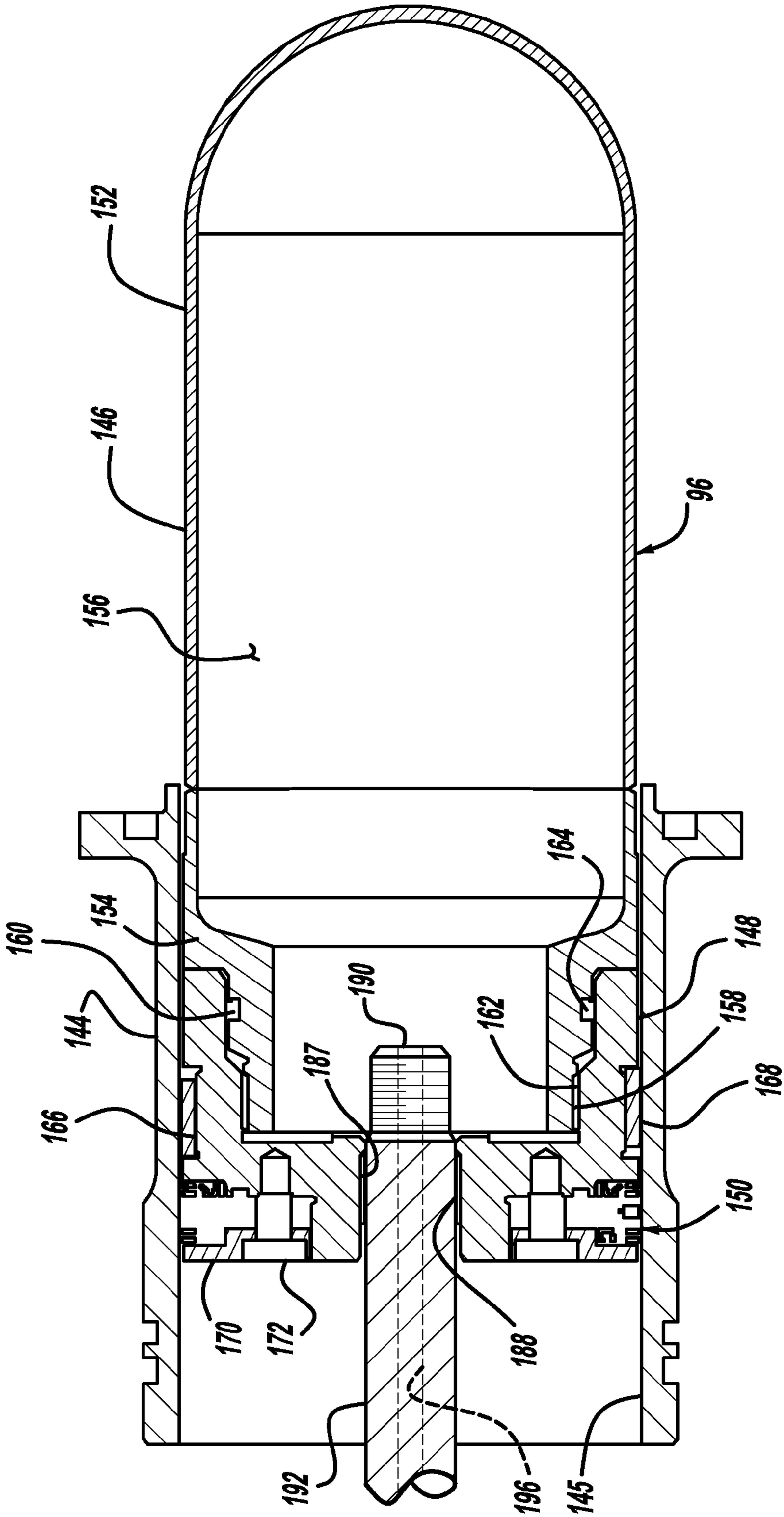


FIG - 6

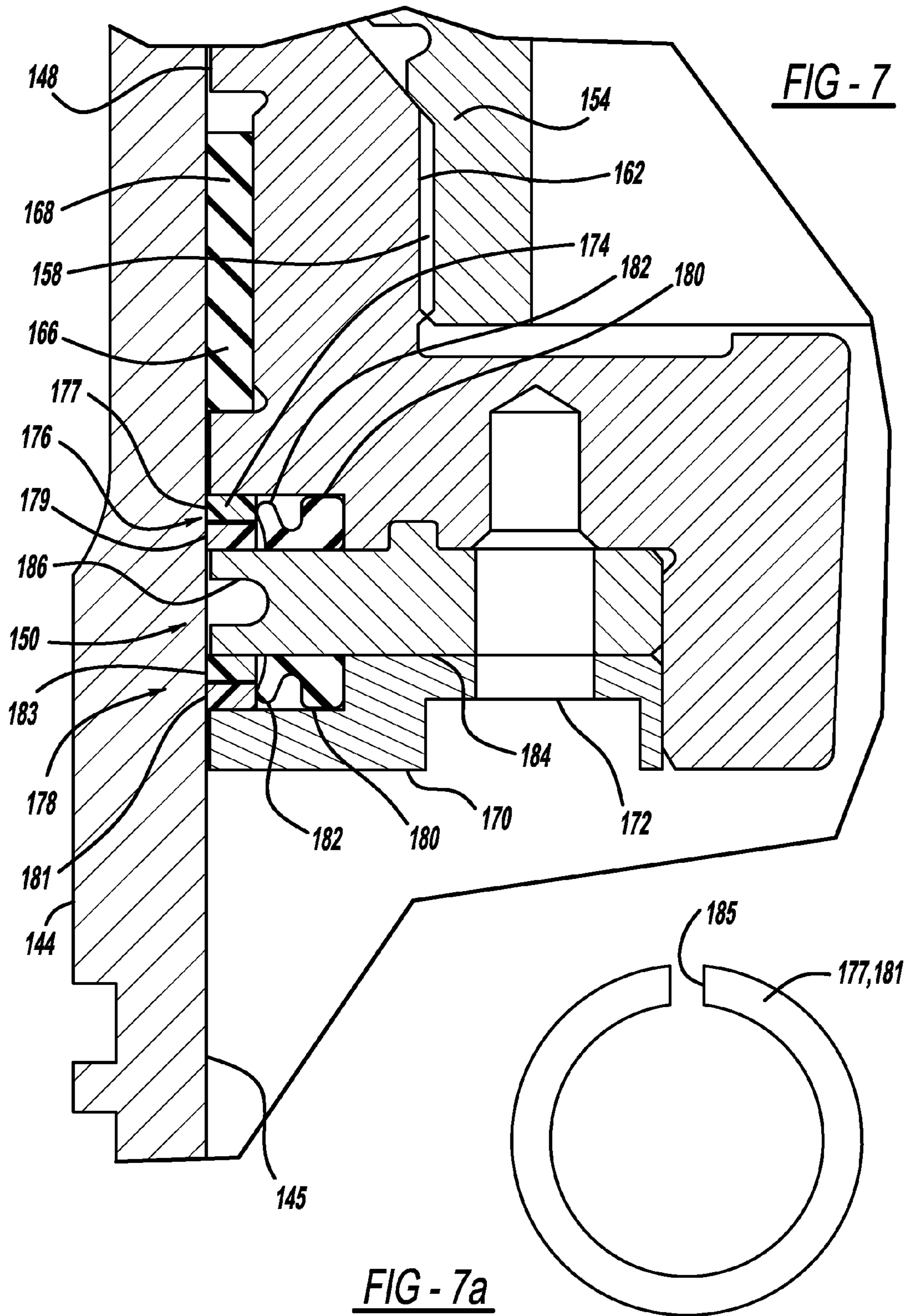


FIG - 7a

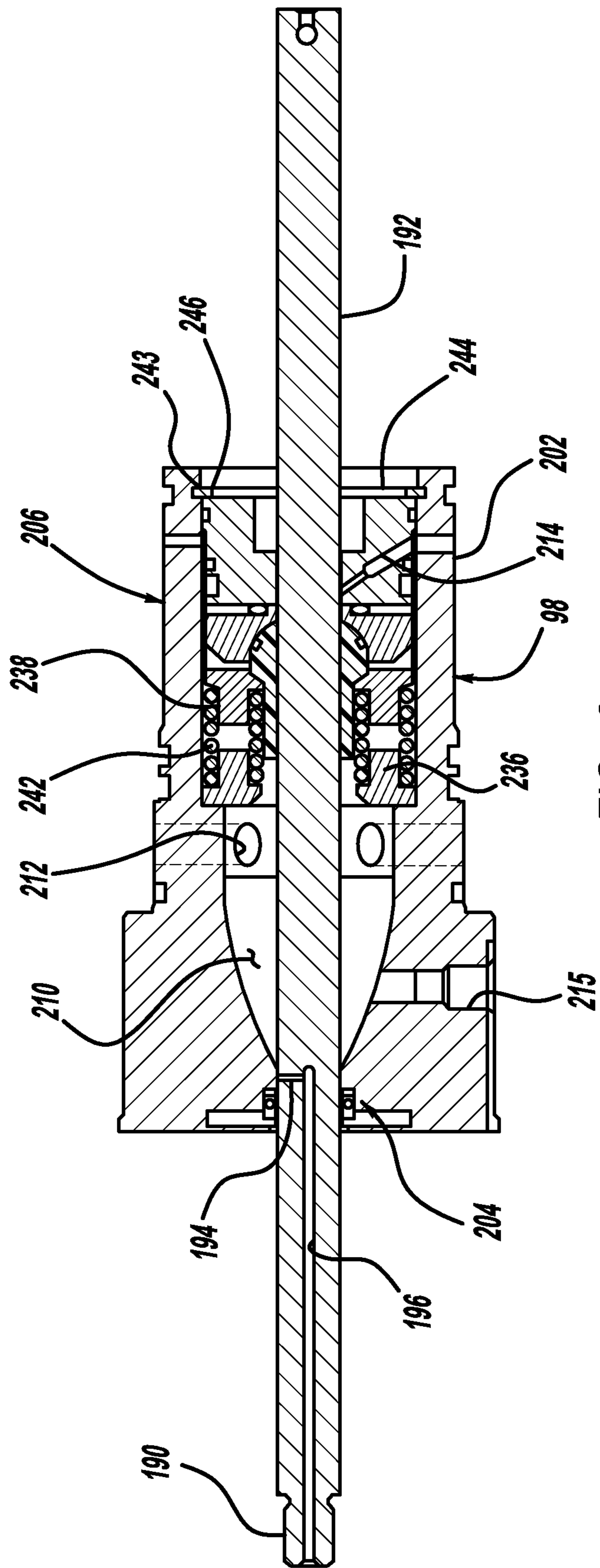


FIG - 8

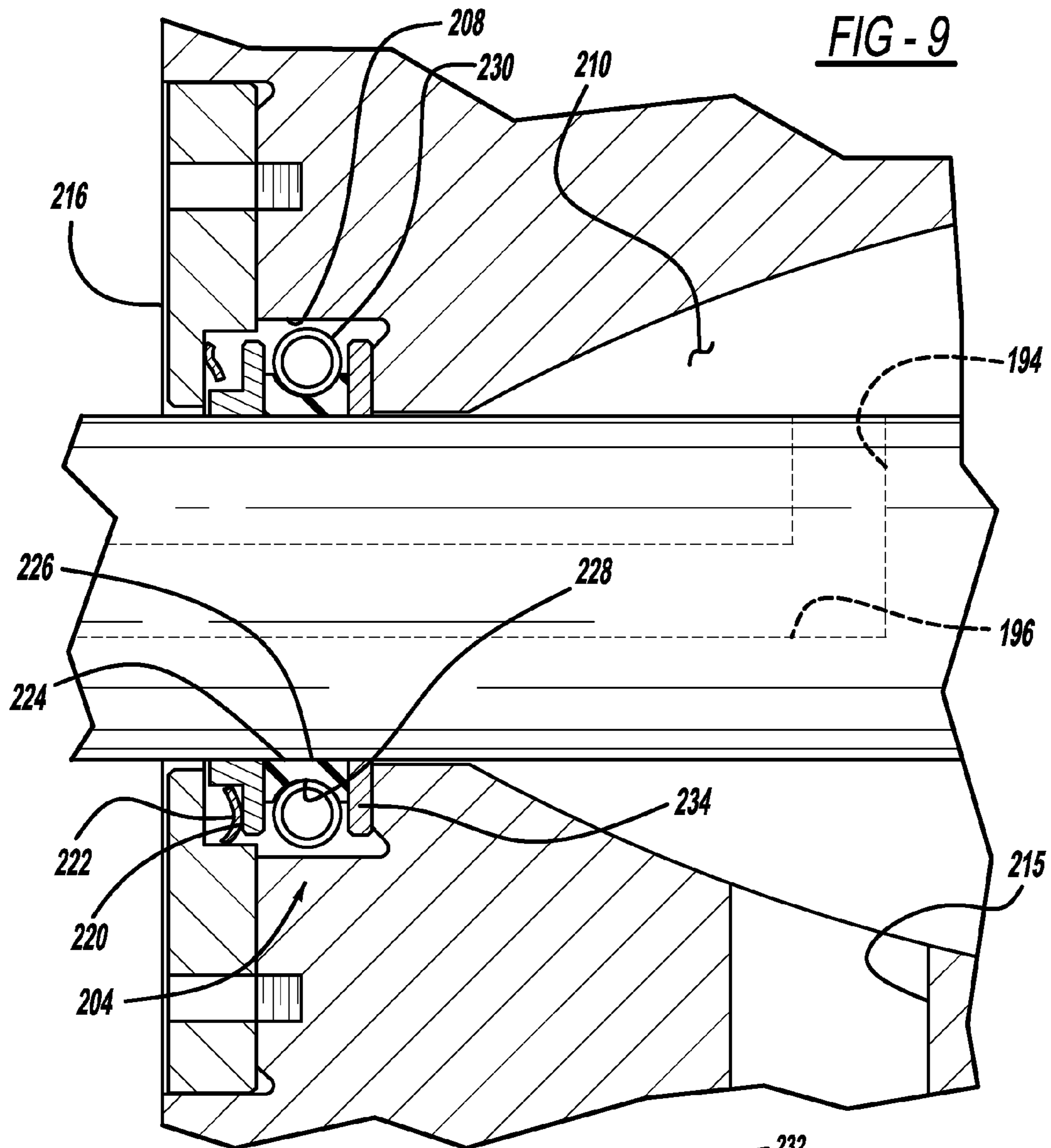


FIG - 9

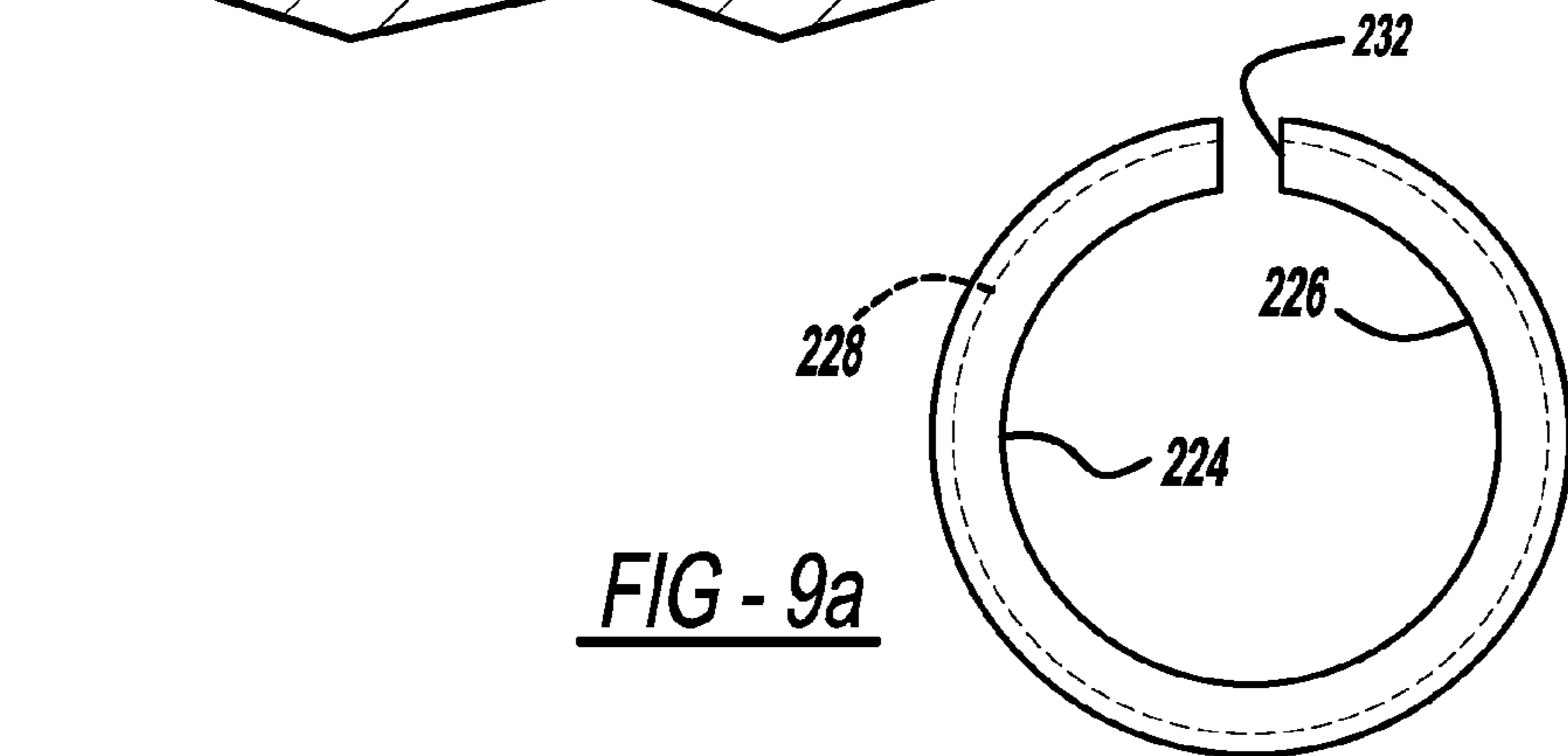


FIG - 9a

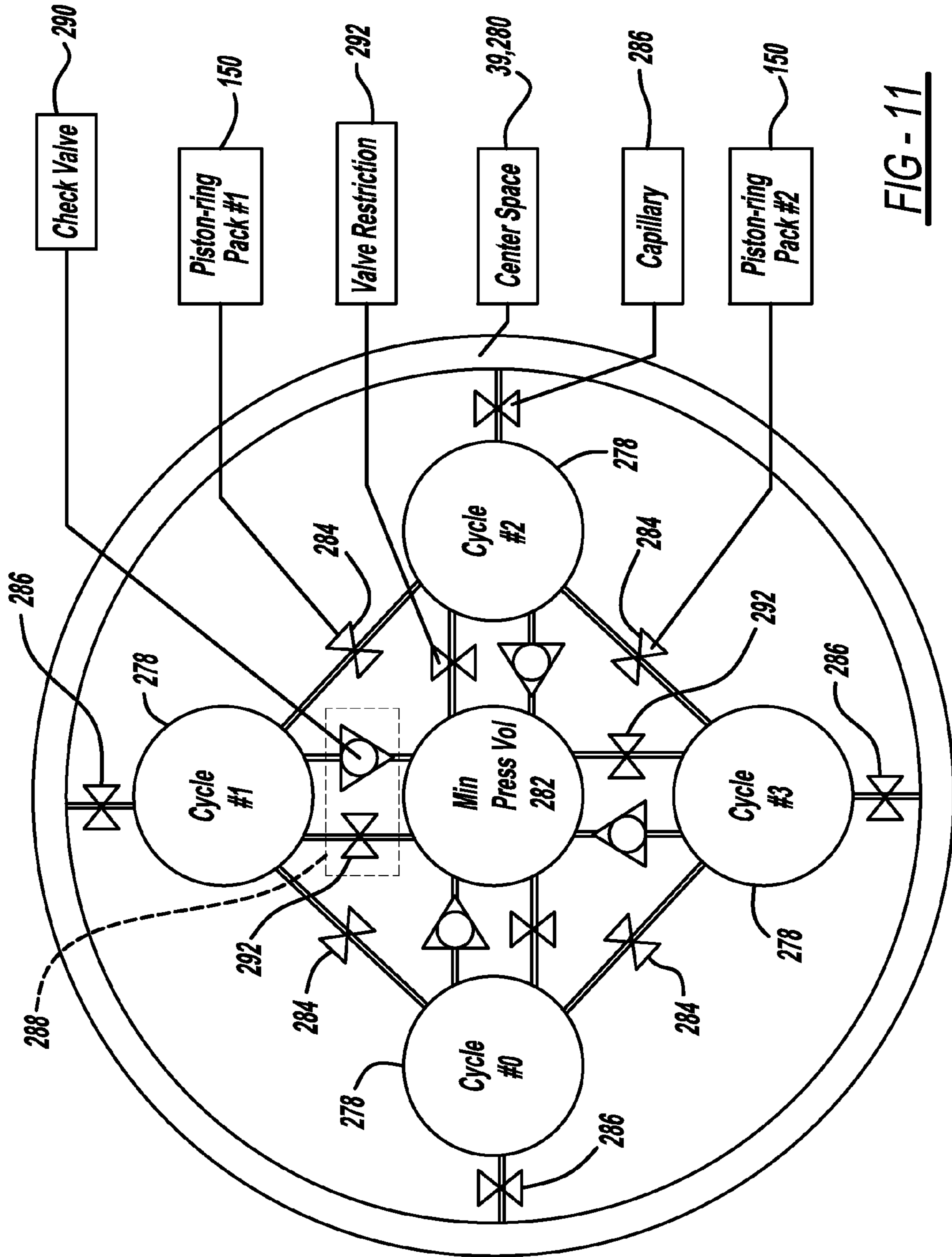


FIG - 11

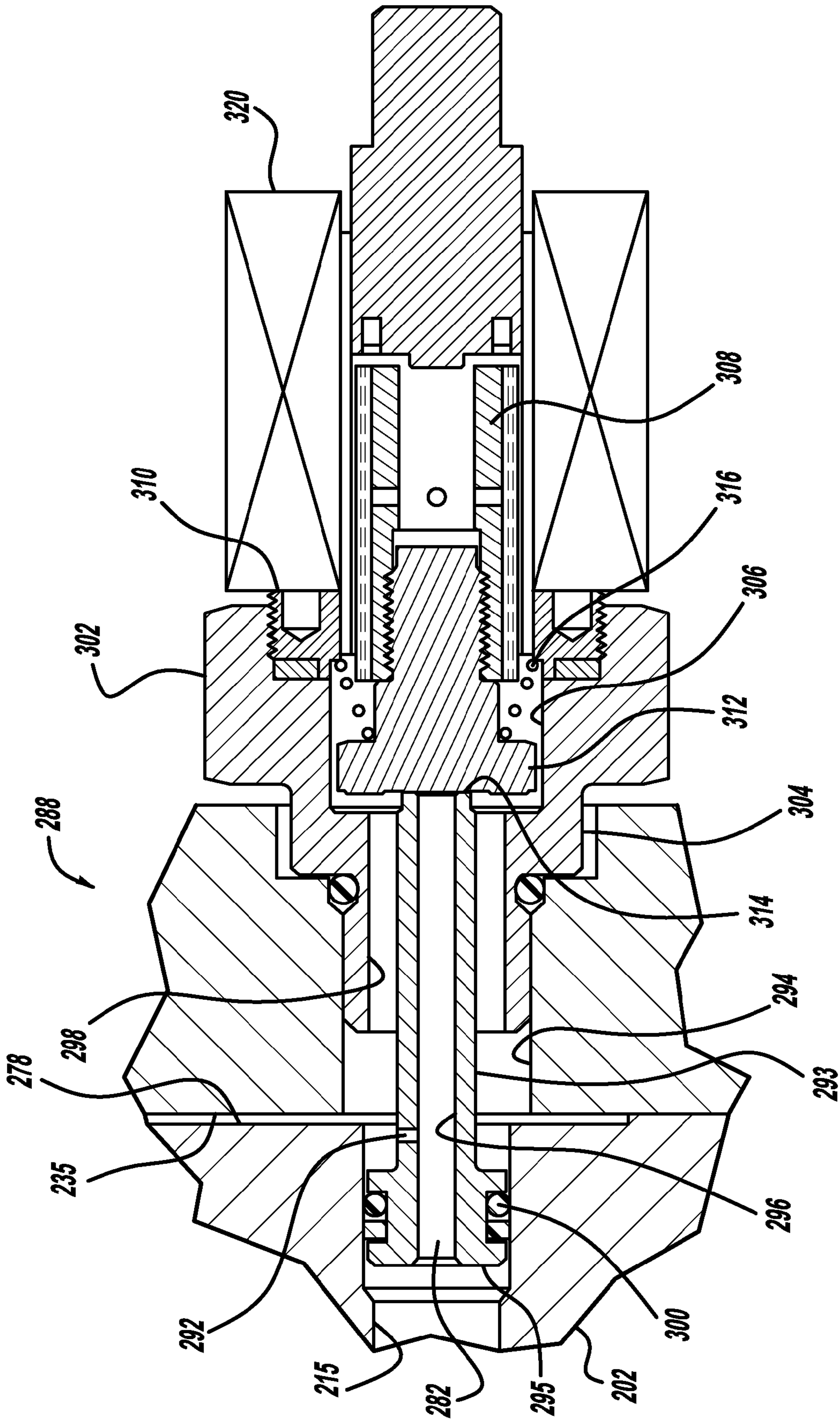


FIG-12

CONTROL VALVE FOR A STIRLING ENGINECROSS REFERENCE TO RELATED
APPLICATION

This application claims priority to provisional patent application No. 61/151,783 filed Feb. 11, 2009.

FIELD OF THE INVENTION

This invention is related to a heat engine and particularly to an improved Stirling cycle engine incorporating numerous refinements and design features intended to enhance engine performance, manufacturability, and reliability.

BACKGROUND AND SUMMARY OF THE
INVENTIONS

The basic concept of a Stirling engine dates back to a patent registered by Robert Stirling in 1817. Since that time, this engine has been the subject of intense scrutiny and evaluation. Various Stirling engine systems have been prototyped and put into limited operation throughout the world. One potential application area for Stirling engines is for automobiles as a prime mover or engine power unit for hybrid electric applications. Other fields of potential use of a Stirling engine such as stationary auxiliary power units, marine applications and solar energy conversion.

Stirling engines have a reversible thermodynamic cycle and therefore can be used as a means of delivering mechanical output energy from a source of heat, or acting as a heat pump through the application of mechanical input energy. Using various heat sources such as combusted fossil fuels or biogases, or concentrated solar energy, mechanical energy can be delivered by the engine. This energy can be used to generate electricity or can be directly mechanically coupled to a load.

The Assignee of the present application, Stirling Biopower, Inc. and its predecessor company have made significant advances in the technology of Stirling machines through a number of years. Although the Assignee has achieved significant advances in Stirling machine design, there is a constant need to further refine the machine, particularly if the intended application is in large volume production.

The Stirling engine of the present invention bears many similarities to those previously developed by Assignee and its predecessor company, including those described in U.S. Pat. Nos. 4,439,169; 4,481,771; 4,532,855; 4,579,046; 4,615,261; 4,669,736; 4,836,094; 4,885,980; 4,707,990; 4,996,841; 4,977,742; 4,994,004; and 5,074,114, which are hereby incorporated by reference. Basic features of many of the Stirling machines described in the above referenced patents are also implemented in connection with the present invention.

The Stirling engine in accordance with the present invention has a so-called "modular" construction. The major components of the engine, comprising the drive case and cylinder block, are bolted together along mating surfaces. Piston rod seals for the pistons traverse this mating plane. A sliding rod seal can be used which is mounted either to the drive case or cylinder block. The rod seal controls leakage of the high pressure engine working gas at one end of the piston connecting rod to atmosphere.

In many past designs of Stirling engines, a large volume of the engine housing is exposed to the high working pressures of the working gas. In accordance with the engine of the present invention, the high pressure working fluid is confined to the extent possible to the opposing ends of the cylinder

bores and the associated heat transfer devices and passage-ways. Thus the high pressure gas areas of the Stirling engine of this invention are analogous to that which is encountered in internal combustion engines, and therefore this Stirling engine can be thought of in a similar manner in terms of consideration for high pressure component failure. This benefit is achieved in the present invention by maintaining the drive case at a relatively low pressure which may be close to ambient pressure, while confining the high pressure working fluid within the cylinder block and the connected components including the cylinder extension, regenerator housing, and heater head.

The pistons of the engine are connected to cross heads by piston rods. The cross heads of the engine embrace the swash-plate and convert the reciprocating movement of the piston connecting rods and pistons to rotation of the swashplate. The Stirling engine of this invention implements a pair of parallel guide rods mounted within the drive case for each cross head. The cross heads feature a pair of journals which receive the guide rods.

The combustion exhaust gases after passing through the heater head of the engine still contain useful heat. It is well known to use an air preheater to use this additional heat to heat incoming combustion air as a means of enhancing thermal efficiency. In accordance with this invention, an air preheater is described which provides a compact configuration with high thermal efficiency.

In the Stirling engine of the type according to the present invention employing four double acting cylinders, there are four discrete volumes of working gas which are isolated from one another (except by leakage across the pistons). In order to enable the engine to operate smoothly and with minimal force imbalances, the mean pressure of each of these four volumes need to be equalized. In accordance with this invention, this is achieved in part by connecting together the four volumes through small orifices. In addition, a system is provided for determining that the mean pressure in each cycle is within a predetermined range. Upon the occurrence of a component failure causing leakage, a significant imbalance could result which could have a destructive effect on the engine. The Stirling engine according to this invention features a pressure control system which unloads the engine upon the occurrence of such failure.

The Stirling engine in accordance with the present invention features a control valve component which, in part, provides the unloading feature mentioned previously. The control valve also provides one of the intended working gas leakage paths which forms part of the pressure balancing system in accordance with the present invention.

A critical component in the Stirling engine of the type described previously involves providing highly reliable seals between the high pressure displacer pistons and the low pressure drive case of the machine. Separating these two volumes is a piston rod seal assembly. Each piston connecting rod reciprocates through a piston rod seal which needs to reliably seal against the piston rod to maintain a low loss rate of working gas to the atmosphere. Absolute sealing of gas leakage through this area is likely not achievable. However, the piston rod seal assembly in accordance with the present invention provides low levels of leakage and reduces contamination of the working gas through "pumping" of lubricating oil in the drive case region.

Another critical design feature for enhancing efficiency of the Stirling engine comes from the design of the piston assembly. The displacer piston separates the hot and cold fluid spaces of the engine and reacts against gas pressures in these areas to deliver mechanical power. Thermal conduction

losses across the piston between the hot and cold spaces need to be minimized to enhance efficiency. Moreover, a highly reliable sliding gas seal is required between the piston rings and the cylinder bore. In addition to constituting a thermal loss, such leakage across the piston seals further results in a net mass exchange of working gases between the individual cycle volumes of the Stirling engine. Significant differences in leakage across the piston seals can result in rapidly changing gas volumes in the cycle volumes. Although means are provided in accordance with this invention for reducing such imbalances, it is desirable to reduce the rate at which these imbalances occur.

Additional benefits and advantages of the present invention will become apparent to those skilled in the art to which this invention relates from the subsequent description of the preferred embodiments and the appended claims, taken in conjunction with the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a longitudinal cross-sectional view through a Stirling engine in accordance with this invention;

FIG. 2 is an enlarged cross-sectional view of the drive case assembly shown in FIG. 1;

FIG. 3 is an enlarged cross-sectional view of the cylinder block section shown in Figure;

FIG. 4 is an enlarged cross-sectional view taken from FIG. 1 showing in greater detail the heater assembly of the engine of this invention;

FIG. 5 is an end view of the cylinder block component taken from the heater assembly end of the engine with the heater assembly omitted;

FIG. 6 is a cross-sectional view of the piston assembly in accordance with the present invention;

FIG. 7 is an enlarged view of the piston seal portion of FIG. 6;

FIG. 7a is an elevational view of a seal shown in FIG. 7;

FIG. 8 is a cross-sectional view of the rod seal assembly in accordance with the present invention;

FIG. 9 is an enlarged cross-sectional view of the cap seal assembly also shown in FIG. 8;

FIG. 9a is an elevational view of a seal shown in FIG. 9;

FIG. 10 is an enlarged cross-sectional view of the base seal assembly shown in FIG. 8;

FIG. 11 is a diagrammatic view showing the pressure balancing circuits of the Stirling engine in accordance with the present invention; and

FIG. 12 is a cross-sectional view of a solenoid operated control valve in accordance with the present invention.

DETAILED DESCRIPTION OF THE INVENTION

Stirling engine in accordance with this invention is shown in an assembled condition in FIG. 1 and is generally designated by reference number 10. Stirling engine 10 includes a number of primary components and assemblies including drive case assembly 12, cylinder block assembly 14, and heater assembly 16 (best shown in FIG. 4).

Overall Construction

Drive case assembly 12 includes a housing 18 having a pair of generally flat opposed mating surfaces 20 and 22 at opposite ends. Mating surface 22 is adapted to be mounted to cylinder block assembly 14. Drive case housing 18 has a hollow interior and includes a journal 24 for mounting a drive shaft bearing. Arranged around the interior perimeter of drive

case housing 18 is a series of cross head guides 26. A pair of adjacent guides 26 is provided for each of the four cross head assemblies 56 of the engine (which are described below). As will be evident from a further description of Stirling engine 10, it is essential that adjacent guides 26 have running surfaces which are parallel within extremely close tolerances.

At one end of drive shaft 40 there is provided journal bearing 24. Drive case housing 18 also provides a central cavity within which oil pump 44 is located. Oil pump 44 could be of various types such as a gerotor type. Through drilled passageways 45, high pressure lubricating oil is forced into spray nozzles which spray a film of lubricant onto the piston rods (described below). In addition, lubricant is forced through internal passages 41 within drive shaft 40 to provide lubrication for swashplate 52.

At a lower portion of drive case 18, a sump port 50 is provided. The lubrication system of engine 10 can be characterized as a sump type with oil collecting in the interior cavity of drive case 12 being directed to oil pump 44 by suction, where it is then pumped to various locations and sprayed as mentioned previously.

Driveshaft 40 supports swashplate 52 which is generally circular and planar but is oriented at an angle inclined with respect to the axis of rotation of the driveshaft. Rotation of driveshaft 40 causes swashplate 52 to rotate about the axis of rotation in an inclined plane. This basic swashplate configuration is a well known design implemented by the Assignee and its predecessor company in prior Stirling engine configurations. Attached to an end of drive shaft 40 is an output coupler 54 enabling connection to a mechanical load, which as previously stated, may be of various types. Flywheel 53 rotates with drive shaft 40. Inductive pickup 55 is positioned near the outer diameter of flywheel 53 and responds to teeth or gaps in the outer diameter to provide electrical signals related to flywheel rotation.

Cylinder block assembly 14 defines a series of four counter bored rod seal bores 48 with passageways 37 and 35 connecting between them (shown in FIG. 5). A number of components attached to mounting surface 49 including regenerator housings 122 and cylinder extensions 116 which are described in more detail in the following sections. Cylinder block assembly 14 further defines four cylinder bores 94 aligned with the rod seal bores 48. Additional components are attached to cylinder block surface 90. Cylinder extension 116 and regenerator housings 122 are connected with cylinder block surface 90 which are bolted to the cylinder block.

Cross Head Assembly

With continued reference to FIGS. 1 and 2, cross head body 58 forms a caliper with a pair of legs 60 and 62 connected by center bridge 64. Each of legs 60 and 62 define surfaces for running along guides 26. Cross head leg 62 also forms slider cup bore 72. Within bore 72 is positioned slider cup 74 which forms a semispherical surface 75. Crosshead leg 60 has a surface machined with a semispherical surface 76. Slider elements 78 and 80 also define spherical outside surfaces which are nested into the mating slider cup surfaces 75 and 76. Opposing flat surfaces 82 and 84 are formed by the slider elements and engaged swashplate 52. The swashplate surfaces may be made conical or crowned to generate line contact with the slider elements.

Cylinder Block

Cylinder block assembly 14, best shown in FIGS. 1 and 3, includes a cylinder block casting 86 having a pair of opposed

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parallel generally flat mating surfaces **88** and **90**. Mating surface **88** enables cylinder block casting **86** to be mounted to drive case housing mating surface **22**. Bolts **92** hold these two parts together. Stirling engine **10** according to the present invention is a four cylinder engine. Accordingly, cylinder block casting **86** defines four cylinder bores **94** which are mutually parallel. As shown in FIG. 1, cylinder bores **94** define a larger diameter segment through which piston assembly **96** reciprocates, as well as a reduced diameter clearance bore section for rod seal assembly **98**. Four cooler bores **102** are also formed in cylinder block casting **86** and are mutually parallel as well as parallel to cylinder bores **94**. Cylinder bores **94** are arranged in a square cluster but lie on a circle outside that of cooler bores **102** as best shown in FIG. 5. In that Stirling engine **10** is a double acting type, cylinder block casting **86** including working gas passageways (not shown) which connect the bottom end of regenerator bore **102** to the bottom end of an adjacent cylinder bore **94**. Cylinder block casting **86** further forms coolant passageways which provide for a flow of liquid coolant through cylinder block **14**. Pressure transducer ports **34** enable mounting of pressure transducers (not shown) for measuring cycle pressure at the bottom of each piston assembly **96**.

Cooler Assembly

Cooler assemblies (not shown) are disposed within cylinder block cooler bores. Cooler assemblies may comprise a shell and tube type heat exchanger with a number of tubes arranged to extend between ends of the housing. The Stirling cycle working gas is shuttled back and forth between the ends of the cooler housing and passed to the inside of the tubes. A coolant, preferably a liquid, is pumped in a cross flow manner through block coolant passages **107** and through the cooler assemblies to remove heat from the working gas.

Cylinder Extensions

Cylinder block assembly **14** further mounts tubular tops or extensions **116** which form a continuation of cylinder block bores **94**. At their open ends, tubular cylinder extensions **116** form a skirt **117** which allows them to accurately align with cylinder bores **94** by piloting. Seals **118** provide a fluid seal between cylinder block bores **94** and tubular cylinder extensions **116**. Cylinder extensions **116** at their opposite ends form a heater tube manifold **120**.

Regenerator Housing

Cup shaped regenerator housings **122** are provided which are aligned coaxially with cooler bores **102**. Regenerator housing **122** define an open end and a closed top **126** having manifold **128** for communication with the heater assembly **16**. Within regenerator housing **122** is disposed a regenerator (not shown) which, in accordance with known regenerator technology for Stirling engines, is comprised of a material having high gas flow as well as high thermal conductivity and heat absorption characteristics.

Heater Assembly

Heater assembly **16** provides a means for inputting thermal energy into the Stirling cycle working gas and is shown in FIG. 4. A combustor (not shown) is used to burn a fossil fuel and other combustible material. Alternatively, heat can be inputted from another source such as concentrated solar energy, or other sources. In Stirling engine **10**, combustion

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gases flow axially toward the center of the engine where it is deflected to outwardly flow in a radial direction. An array of heater tubes **136** is arranged to conduct heat from the high gases that flow radially out of the engine. Heater tubes **136** are arranged to form an inner band and outer band with heat absorbing fins **140** therebetween. Heater tube manifolds **120** and **128** have internal passageways which connect the inner and outer bands of heater tubes **136**.

Air Preheater

Combustion gases which pass through heater tubes **136** are still at an elevated temperature and have useful heat energy which can be recovered to enhance the thermal efficiency of engine **10**. This is achieved through the use of an air preheater (not shown) which has an annular ring configuration and surrounds the outer bank of heater tubes **136**. The air preheater transfers waste heat from the exhaust gases.

Piston Assembly

Piston assembly **96** is shown in assembled condition in FIG. 6. As illustrated, piston assembly **96** principally comprises piston dome **146**, piston base **148**, and ring assembly **150**. Piston assembly **96** slides within cylinder liner **144** which is mounted within cylinder block bores **94**.

Piston dome **146** forms a hollow dome top **152** and a machined base section **154**. Dome top **152** has a hollow interior **156**. Since piston assembly **96** separates hot and cool spaces formed by the working gas, it is desirable to minimize heat exchange between the top and bottom ends of piston assembly **96** (with the top end at the right hand side of FIG. 6 and the bottom end at the left hand side). Heat transfer through piston assembly **96** between its ends represents a thermal efficiency loss for Stirling engine **10**. Accordingly, dome top **152** has a thin wall construction to minimize conductive heat transfer and interior **156** is hollow to minimize conductive heat paths. Piston base section **154** has a machined outer stepped diameter. The end of base section **154** forms threaded section **158** for assembly with piston base **148**.

Piston base **148** has a machined bore section forming internal threaded section **162** which meshes with external threaded section **158** to enable piston dome **146** and base section **154** to be threaded together and assembled as shown in FIG. 6. Seal **164**, located in seal groove **160**, provides sealing connection between these components.

Referring also to FIG. 7, the outer surface of piston base **148** features bearing groove **166** which receives elastomeric piston bearing **168**. Bearing **168** has a thickness sufficient to bear directly against the inside surface (or bore) **145** of cylinder liner, keeping the metal components of piston assembly **96** from directly contacting the inside bore surface of cylinder liner **144**. Bearing **168** does not provide gas sealing with bore **145** (i.e., it acts as a bearing and not a seal). The lower end of piston base section **148** provides a region for installation of piston ring assembly **150** which is sandwiched in place by fastening piston base plate **170** using cap screws **172**. Ring assembly **150** is shown in greater detail in FIG. 7.

Ring assembly **150** is fit within annular groove **174**. Ring assembly **150** forms a pair of ring sets including upper ring set **176** and lower ring set **178**. Ring sets **176** and **178** are positioned symmetrically on opposite radial surfaces of piston base land **184** which is held into position through clamping engagement between piston base plate **170** and piston base **148**. Piston base land **184** is preferably formed of steel material and features an annular outer groove **186**. Upper ring set **176** includes rings **177** and **179**. Lower ring set **178** includes

rings **181** and **183**. The rings of ring sets **176** and **178** are preferably formed of an elastomeric material. Expander rings **180**, also formed of an elastomeric material, are placed radially inboard of the ring sets **176** and **178**, and have a sealing lip **182** which bears against ring **177** of the upper set and **181** of the lower set. Rings sets **176** and **178**, expander rings **180**, and bearing **168** may all be formed from a PEEK (polyetheretherketone) material.

Ring **177** of upper ring set **176** and ring **181** of lower ring set **178**, both illustrated in FIG. **7a** feature a radial split **185** which provides a small leakage path for working gas to pass across the rings in a controlled manner. Rings **179** and **183** however are solid and do not feature a radial split. The radial splits **185** of rings **177** and **181** prevent pressure imbalances from occurring which otherwise can lead to pumping conditions.

Piston base section **148** forms a central tapered bore **188**. Piston rod **192** has a tapered upper end **187** which fits within tapered bore **188**. Piston rod **192** may have a threaded end **190** which can be engaged by an assembly tool to press-fit the piston rod tapered end into piston base taper bore **188**. Piston rod **192** can be press-fit into tapered bore **188** and, after that operation is completed, these parts are connected. Piston dome **146** may be threaded in an assembled condition with base **154**.

Rod Seal Assembly

Rod seal assembly **98** is best shown with reference to FIGS. **8**, **9**, and **10**. As illustrated, piston rod **192** is shown passing through rod seal assembly **98**. As illustrated, piston rod **192** includes a central drilled bore **196** extending from its threaded end **190** to a point within rod seal assembly **98** intersecting with radial passageway **194**. Passageways **196** and **194** communicate with piston dome interior **156** and to a hollow interior cavity **210** of the rod seal assembly **98** with the cavities maintained at a cycle minimum pressure, as described previously. Rod seal assembly **98** includes additional principal components including housing **202**, cap seal assembly **204**, and base seal assembly **206**.

Housing **202** is formed of a rigid material such as steel and forms a recessed stepped bore **208** at its end facing piston assembly **96**. Interior bore **210** forms a hollow interior volume and a stepped bore for accommodating base seal assembly **206**. Radial passageways **212** are provided to communicate the interior cavities **210** of each of four rod seal assemblies **98** through passageways **37** drilled into the block. Interior cavities **210** are connected together with piston cavities **156** and define a volume at near minimum cycle pressure, designated later in this description as volume **282** in FIG. **12**. Radial passageway **214** provides a flow passageway for lubricating oil to be sprayed against the outer surface of piston rod **192** for providing lubrication and cooling. The outer diameter of housing **202** features a number of grooves and locating features to enable it to be sealed and mounted in position within rod seal bores **48** of block **14**.

Cap seal assembly **204** is shown in detail in FIG. **9** and is maintained in position by fastening cap seal plate **216** using screws. Cap seal assembly **204** is trapped within the hollow interior cavity provided when cap seal plate **216** is fastened and positioned. As shown, at its end closest to piston assembly **96**, cap seal assembly includes cap seal spacer **220** having an L-shaped cross-sectional configuration. Wave springs **222** are loaded into the radial cavity formed by cap seal spacer **220** and are used to place an axial load against other components of cap seal assembly **204**. Cap seal spacer **220** bears against cap seal **224** which has an inner diameter **226** which bears

against the outer surface of piston rod **192**. Cap seal **224** has a semi-circular outer groove **228** and coil spring **230** is positioned around cap seal **224** and places a radially inward compressive force on the cap seal for enhanced sealing against the piston rod. Cap seal **224** features a radial split **232** which provides an intended gas leakage path. Cap seal **224** bears against cap seal seat washer **234**. Cap seal assembly **204** is used to provide a wiping function along the outer diameter of piston rod **192**. This provides a reduction in the pressure variation across cap seal assembly **204** to provide more effective gas sealing for the remaining components of the seal assembly **98**. Cap seal assembly **204** further provides an oil wiping function to remove lubricating oil from the outer surface of piston rod **192** to prevent contamination of the cycle gases.

Base seal assembly **206**, best shown in FIG. **10**, is positioned within housing interior bore **210**. A pair of annular spring seats **236** and **238** is provided which produce an axial loading force for base seal assembly **206**. Spring seats **236** and **238** feature posts **240** and **241** which locate coil springs **242**. As illustrated, spring seat **236** bears against a shoulder within interior bore **210** to provide the axial loading force. Seal plug **244** is located in position within the housing bore and is maintained there through a snap ring **246** installed within groove **248**. Seal plug **244** further features passageway **214** for the transfer of lubricating oil to spray against piston rod **192**. The radial end face surface **254** of plug **244** is polished. Bearing against it is washer **256** made from a polyimide material and inner O-ring **258**. Washer **256** and O-ring **258** permit seat **260** to shift relative to plug **244** with low friction. This allows seal seat **260** to stay centered up with rod **192** during operation of the engine **10**.

Seal seat **260** features a concave semi-spherical surface **262** and a projecting inner post **264** which traps O-ring **258** in this trapped position. Rod seal **266** forms a projecting tubular section **268** and head **270** forming a convex semi-spherical surface **272** which further forms seal groove **274** which retains seal **276**. Ideally, seal groove **274** is formed such that its side surfaces are tangent to the semi-spherical surface **272** of seal **266**. The semi-spherical surface **262** of the seal seat **260** and its mating surface **272** of seal **266** enable the seal to respond to bending of piston rod **192** during operation of engine **10**, as well as adjust to any misalignments of the piston rods with respect to rod seal assembly **98**. The elastomeric components of cap seal assembly **204**, including cap seal **224** and rod seal **266** may be formed from a PTFE (polytetrafluoroethylene) material. Although surfaces **262** and **272** are described as "semi-spherical", other matching concave and convex shapes deviating from purely spherical may also be used.

Pressure Equalization System and Control Valve Assembly

In operation of Stirling engine **10**, it is important that the total mass of working fluid contained in the four enclosed volumes of the engine are maintained to have closely equal masses of working fluid. This is necessary to prevent average pressure differences between the enclosed volumes and therefore force imbalances from occurring in the engine. There are unavoidable losses of working gas through heater head assembly **16** and other leakage paths, as well as minute leakage across piston assembly **96** and across rod seal **98**. Consequently, the Stirling engine in accordance with this invention provides a mechanism for allowing equalization in the mass of working gas existing in the four separate cycle volumes (each bounded at the top of one piston assembly **96** and

the bottom of an adjacent piston assembly). In addition, it is desirable to reduce the starting torque required acting on driveshaft 40. This enables smaller capacity starting motors having lower torque outputs to be used for starting the engine. These systems are best described with reference to FIGS. 11 and 12. Cycle volumes of working gas 278 (typically hydrogen and helium) are designated in the figures as cycle volumes #0, #1, #2, and #3.

FIG. 11 shows diagrammatically a system providing pressure balance. As the diagram illustrates, the gas of each of the cycles are represented in the figure as cycle volumes 278 designated individually as "cycle #0", etc. These cycle volumes are interconnected through a number of pathways. Two pressure volumes are formed within engine 10 including passageways 37 drilled into cylinder block 14 in a square arrangement when viewing the engine as shown in FIG. 5 which are connected with rod seal internal cavities 210 and piston interiors 156 to collectively form a minimum pressure volume 282. Passageways 35 having restrictors 286, which may be in the form of capillary tubes having a diameter for example of 0.4 mm, communicate with the working fluid space formed at the bottom of piston assemblies 96 and are therefore exposed to cyclically varying cycle pressure. The internal volume of passageways 35 at their point of intersection past restrictors 286 in the center space 39 of the engine form a mean pressure volume 280.

As stated previously, a minute leakage occurs between cycle volumes #0, #1, #2, and #3 across the rings of piston assemblies 96. This leakage pathway is designated diagrammatically in FIG. 11 as restrictor 284. Passageways 35 drilled within cylinder block 14 provide a common volume through which cycle leakage occurs through restrictors 286. This allows a small net flow of working gas to be maintained within the mean pressure volume 280. Since the pressure applied to restrictors 286 cycles between the cycle volumes 278 maximums and minimums, a small net flow periodically occurs in both directions through the restrictors and thus maintaining volume 280 at near mean cycle pressure (here "mean" refers not only to a pressure which is the average of the minimum and maximum pressures, but to any intermediate pressure between the cycle minimums and maximums). Restrictors 286 are represented diagrammatically in FIG. 11. As stated previously, rod seal volumes 210 are held a minimum pressure volume 282. Valve port 294 is exposed to minimum cycle pressure volume 282 through rod seal passageway 215, and to cycle pressure 278 in a radial space 235 surrounding rod seal assembly 98 (connected together through passageways 35 and restrictors 286). Housing ports 294 are aligned with cap seal passageway 215 in cap seal housing 202. A radial space 235 between cap seal housing 202 and the cylinder block 86 communicates with the bottom of the pistons 96 and therefore undergoes the cyclically varying gas pressure of the working gas volumes.

Valve assemblies 288 are provided for each of the cycle volumes and are described in more detail as follows. Diagrammatically, valve assemblies 288 act as a solenoid actuated check valve 290. Valve assembly 288 also produces a leakage path through valve orifice 292 acting as a restrictor. When valve 288 is electrically actuated, a free flow between the cycle volumes 278 into minimum pressure volume 282 occurs. This minimizes engine starting torque and allows piston assemblies 96 to be reciprocated with low starting torque.

Valve 288 is illustrated in detail in FIG. 12. Valve assemblies 288 are each installed within ports 294 in cylinder block 86 with plug 293 extending into cap seal passageway 215. As illustrated in FIG. 11, valve assemblies 288 control fluid

movement between the minimum pressure volume 282 and the cycle volumes 278. These two pressure volumes are separated through sealing provided by O-ring 300 around the head end 295 of plug 293. Valve assembly 288 includes valve body 302 having a threaded end 304, allowing it to be fixed into position within the associated ports 294. Valve body 302 forms an internally stepped bore 306. Sleeve assembly 308 is fastened in position relative to valve body 302 by installing threading cap 310. Within sleeve assembly 308 is moveable plunger 312 which is held in a normal position against seat 314 by coil spring 316. Coil winding 320 surrounds sleeve assembly 308. When electric current is passed through winding 320, plunger 312 is caused to move away from seat 314, allowing free passage of fluid between the volumes 282 and 278, thus effectively connecting together the cycle volumes. This free passage of gas between cycles reduces starting torque and can rapidly reduce power output in a condition of a failure of an engine component or other need to quickly unload the engine. Since plunger 312 is spring loaded into engagement with seat 314, higher pressure in passageway 296 urges plunger 312 to move away from sealing engagement with seat 314 and thus the valve assembly 288 acts as a check valve 290 in conditions where current is not flowing into coil windings 320. In one embodiment, valve 288 has a plunger lift-off (or cracking) pressure of less than about 1.0 Mpa (i.e., plunger 312 unseats at that pressure difference). Unless plunger 312 is actuated, gas flow in a reverse direction (from cycle volumes 278 to minimum pressure valve 282) is inhibited (although a controlled "leak" occurs across restrictor 292).

Referring back to FIG. 11, whenever the minimum pressure volume 282 is not greater than the lowest pressure occurring in the cyclical pressure variations of any of the cycle volumes 278 by more than the lift-off pressure of plunger 312, no fluid is transferred through valve assembly 288. If however, the minimum pressure experienced in any one of the cycle volumes 278 is less than the pressure of minimum pressure volume 282 by more than the lift-off pressure, a net force acts on valve assembly plunger 312 to urge it to open. A spring force applied to plunger by coil spring 316 is adjusted such that if this difference in pressure exceeds a predetermined amount (the lift-off pressure), plunger 312 unseats allowing fluid to be moved out of minimum pressure volume 282, thus maintaining at its desired low pressure value in that volume. Orifice 292, shown in FIG. 12 and diagrammatically in FIG. 11 provides a controlled leak between the cycle volumes 278 and minimum pressure volume 282. This mechanism is another way in which gas is exchanged between the cycle volumes 278 which maintains the mass of gas in each cycle #0, #1, #2, and #3 to be balanced as engine 10 is operating.

As mentioned previously, if valve assembly 288 is actuated, the "short circuit" or free flow of gas of cycle volumes 278 to one another interrupts the thermodynamic cycle of operation of Stirling engine 10 but permits low starting torque to put the mechanical components of the engine in motion, and also provides the unloading feature mentioned previously. During a prolonged period after stopping operation of engine 10, the various pressure volumes in the engine with tend to equalize in pressure. Once engine 10 is operated and valve assembly 288 is de-energized (allowing plunger 312 to seat) the cycle volumes 278 undergo their pressure variations from a minimum to a maximum level in a cyclical manner. As mentioned previously, whenever any one of cycle volumes 278 goes to a pressure level less than the existing pressure in minimum pressure volume to 282 by an amount exceeding the check valve lift-off pressure, minimum pressure volume

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282 is “pumped down” to a steady state pressure which is slightly greater than the minimum pressure experienced in the cycle volumes 278. Thus during operation, if any one of the cycles #0, #1, #2, or #3 exhibits a pressure imbalance with the other cycles in which its minimum pressure during cyclical variation is below that of the other cycles, a net flow of working gas into that cycle will occur as its respective check valve 290 operates. The continuous leakage paths provided by each of restrictors 292 causes a net periodic flow through each of the restrictors which is another means by which the volume or mass of working gas in each of the cycle volumes 278 are equalized during operation of the engine. Another mechanism for the exchange of gas between the cycle volumes 278 occurs by the leakage path to the center space 39 of the engine which is held at the mean pressure as mentioned previously. A constant shuttling of minute quantities of gas occurs through each of the restrictors 286 during operation of the engine. Since the mean pressure volume 280 communicates with each of the cycle volumes 278, this mechanism provides a means of exchanging gas between the cycle volumes. It is acknowledged that any leakage of working gas in cycles 278 has the effect of reducing the magnitude of maximum and minimum pressures which leads to an efficiency penalty for the engine. However, by maintaining the leakage gas through restrictors 288 and 286 to minute levels, any degradation in performance becomes negligible.

It is to be understood that the invention is not limited to the exact construction illustrated and described above, but that various changes and modifications may be made without departing from the spirit and scope of the invention as defined by the following claims.

The invention claimed is:

1. A control valve for a double acting type Stirling cycle engine of the type having two or more piston assemblies reciprocating within cylinder bores, each of the piston assemblies separating isolated cycle volumes of a working gas contained within the engine, the cycle volumes undergoing cyclical pressure changes from a minimum cycle pressure to a maximum cycle pressure, the engine having a valve port for connection with the control valve which communicates with one of the cycle volumes and with a pressure volume, the control valve comprising:

a valve body forming a first passageway for communicating with the cycle volume and a second passageway for communicating with the pressure volume,

a plunger moveable between a closed position against a seat which seals against the seat and an open position displaced from the seat, wherein, in the closed position, flow between the first passageway and the second passageway is restricted and in the open position the flow between the passageways is enabled,

a spring urging the plunger to the closed position such that the plunger moves to the open position when the pressure in the first passageway is less than the pressure in the second passageway by more than a predetermined pressure difference, and

an actuator acting on the plunger to move the plunger to the open position against the urging force of the spring to thereby connect the passageways.

2. A control valve in accordance with claim 1 wherein the valve body includes a plug having a bore that forms the second passageway with an end of the plug forming the seat.

3. A control valve in accordance with claim 2 wherein the valve body first passageway surround the plug.

4. A control valve in accordance with claim 2 wherein the plug forms a third passageway which forms a restricted flow connection between the first and second passageways.

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5. A control valve in accordance with claim 2 wherein the actuator is an electrical solenoid.

6. A control valve for a double acting type Stirling cycle engine of the type having two or more piston assemblies reciprocating within cylinder bores, each of the piston assemblies separating isolated cycle volumes of a working gas contained within the engine, the cycle volumes undergoing cyclical pressure changes from a minimum cycle pressure to a maximum cycle pressure, each of the piston assemblies connected with a connecting rod, and having a rod seal assembly for sealing the connecting rod, the engine having a valve port for receiving the control valve which communicates with the cycle volumes and with a pressure volume, the control valve comprising:

a valve body forming a first passageway for communicating with the cycle volume,

a valve plug having a hollow interior passage forming a second passageway for communicating with the pressure volume,

a plunger moveable between a closed position against a seat which seals against the seat and an open position displaced from the seat, wherein, in the closed position, flow between the cycle volume and the pressure volume is restricted and in the open position the flow between the cycle volume and the first volume is enabled,

a spring urging the plunger to the closed position such that the plunger moves to the open position when the pressure in the first passageway is less than the pressure in the second passageway by more than a predetermined pressure difference, and

an actuator acting on the plunger to move the plunger to the open position against the urging force of the spring to thereby connect the passageways.

7. A control valve in accordance with claim 6 wherein an end of the plug forming the seat.

8. A control valve in accordance with claim 6 wherein the valve body first passageway surround the plug.

9. A control valve in accordance with claim 6 wherein the plug forms a third passageway which forms a restricted flow connection between the first and second passageways.

10. A control valve in accordance with claim 6 wherein the actuator is an electrical solenoid.

11. A double acting type Stirling cycle engine comprising: two or more piston assemblies reciprocating within cylinder bores, each of the piston assemblies separating isolated cycle volumes of a working gas contained within the engine, the cycle volumes undergoing cyclical pressure changes from a minimum cycle pressure to a maximum cycle pressure,

the engine having block forming a valve port for each of the cycle volumes which each communicate with one of the cycle volumes and with a pressure volume, and

a control valve disposed in each of the valve ports and forming a first passageway for communicating with the cycle volume and a second passageway for communicating with the pressure volume, the control valve further having a plunger moveable between a closed position against a seat which seals against the seat and an open position displaced from the seat, wherein, in the closed position, flow between the first passageway and the second passageway is restricted and in the open position the flow between the passageways is enabled, a spring urging the plunger to the closed position such that the plunger moves to the open position when the pressure in the first passageway is less than the pressure in the second passageway by more than a predetermined pressure difference, an actuator acting on the plunger to

move the plunger to the open position against the urging force of the spring to thereby connect the passageways.

12. A Stirling engine in accordance with claim **11** wherein, in operation of the engine the control valve acts to cause the pressure in the engine pressure volume to have a pressure close to the minimum cycle pressure. 5

13. A Stirling engine in accordance with claim **11** wherein each of the control valve pressure volumes are connected with the pressure volume.

14. A Stirling engine in accordance with claim **11** wherein the control valve has a second passageway with provides a restricted flow path between the first and second passageways thereby allowing working gas to pass between the cycle volumes and the pressure volume. 10

15. A Stirling engine in accordance with claim **11** wherein when the control valve opens, the cycle volumes are connected with the pressure volume which reduces the magnitude of the difference between the minimum and maximum cycle pressures. 15

16. A Stirling engine in accordance with claim **11** wherein engine includes rod seal assemblies sealing connecting rods coupled with pistons, the rod seal assemblies having a housing with a hollow interior cavity which communicates with the pressure volume. 20

17. A Stirling engine in accordance with claim **11** wherein the engine port communicating with the rod seal interior cavity. 25

18. A Stirling engine in accordance with claim **17** the rod seal internal cavity communicating with a seal port aligned with the valve port and the control valve having the plug with a head positioned in the seal port. 30

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